CONDENSATION HEAT TRANSFER COEFFICIENT CORRELATION BASED ON SLIP RATIO MODEL IN A HORIZONTAL HEAT EXCHANGER

Seok Kim, Sung Uk Ryu, Seung Tae Lee, Dong-Jin Euh, and Chul-Hwa Song
Korea Atomic Energy Research Institute
111 Daedeokdaero989Beon-gil, Yuseong-gu, Daejeon, Korea
seokim@kaeri.re.kr; rsu@kaeri.re.kr; leest@kaeri.re.kr; djeuh@kaeri.re.kr; chsong@kaeri.re.kr

ABSTRACT

Over the past few decades, there have been many experimental and analytical researches for predicting the horizontal condensation heat transfer and numerous models have been proposed. However, the proposed models that were based on the limited database, showed a considerable deviation against the experimental data, depending on authors, working fluids and flow conditions. Furthermore, since the PAFS (Passive Auxiliary Feedwater System) is a unique passive safety system adopted in the APR+ (Advanced Power Reactor Plus), conventional models showed inaccurate prediction capability on the condensation heat transfer coefficient. Experimental investigation of the condensation heat transfer and natural convection phenomena in the PAFS was experimentally investigated at KAERI (Korea Atomic Energy Research Institute). It was found that a thermal hydraulic system analysis code, MARS-KS (Multi-dimensional Analysis of Reactor Safety), underestimated the condensation heat transfer coefficient compared to the experimental data. The current study suggested a condensation heat transfer coefficient correlation using the slip ratio to model the heat transfer phenomenon inside the horizontal condensation heat exchanger. In this study, the modified correlation using the analytic slip ratio model was evaluated with the experimental data. The proposed model showed an improved prediction performance having around 8% of the error. To increase the prediction capability of the condensation heat transfer coefficient correlations developed in this study, more experimental data in the wide ranges should be compared as the further study.

KEYWORDS
Condensation, Heat transfer coefficient, Slip ratio, Passive auxiliary feedwater system, Horizontal heat exchanger,

1. INTRODUCTION

The APR+ (Advanced Power Reactor Plus) adopts several new safety features. The PAFS (Passive Auxiliary Feedwater System) is one of the advanced safety features, which can cool down the nuclear reactor without any external power supply in case of accidents. The PAFS is driven by the natural circulation to remove the core decay heat through the PCHX (Passive Condensation Heat Exchanger) that is composed of nearly horizontal tubes.

For validation of the cooling and operational performance of the PAFS, PASCAL (PAFS Condensing Heat Removal Assessment Loop) facility was constructed and experimental investigation of the condensation heat transfer and natural convection phenomena in the PAFS was experimentally investigated at KAERI (Korea Atomic Energy Research Institute). It was found that a thermal hydraulic system analysis code,

MARS-KS (Multi-dimensional Analysis of Reactor Safety), underestimated the condensation heat transfer coefficient compared to the experimental data.
Over the past few decades, there have been many experimental and analytical researches for predicting the horizontal in-tube condensation heat transfer and numerous models have been proposed. However, the proposed models that were based on the limited database, showed a considerable deviation against the experimental data, depending on authors, working fluids and flow conditions. Moreover, there is not a single model that can be applied to this unique PAFS. Therefore, in order to improve the prediction capability of the condensation heat transfer coefficient and to model the condensation phenomenon inside the PCHX of the PAFS properly using the system analysis code, it is necessary to develop the condensation heat transfer coefficient model that has the improved prediction capability.
The condensation heat transfer coefficient correlation using the slip ratio for the horizontal condensation tube was developed to model the heat transfer phenomenon inside the condensation heat exchanger. In this study, the improved correlation using the analytic slip ratio model was evaluated with experimental data that were acquired from the HOCO (Horizontal Condensation Two-Phase Flow Test Loop) and PASCAL facility [11.12] at KAERI.

2. CONDENSATION HEAT TRANSFER COEFFICIENT CORRELATION

Numerous horizontal in-tube condensation models are available in the literatures, and most of these models can be categorized as shear dominated annular flow or gravity dominated stratified flow models. Park et al. well summarized the previous research works [1] and also proposed the mechanistic correlation for the condensation heat transfer coefficient.
During condensation inside horizontal tubes, the two-phase flow pattern may be dominated by vapor or gravity forces. While annular flow pattern is associated with high vapor shear, stratified flows appear when gravity is the dominant force. In the fully developed annular flow pattern, there is a thin uniform condensate film on the entire tube wall, while the gas phase flows in the central core, and heat transfer is governed by vapor shear and turbulence. In the stratified flow regime, a certain thickness of the condensate layer forms at the bottom of the tube and a thin liquid film settles on the wall in the upper portion of the tube. In this condition, the heat transfer through the thin film is generally analyzed by the classical Nusselt theory. Table I showed the representative correlations for the condensation heat transfer coefficient.
Chato [2] studied the stratified flow with lower vapor velocity. He developed a modified version of the classic Nusselt [3] theory for the falling film condensation. In his analytical model, the heat transfer through the thick condensate layer at the bottom portion of the tube is considered negligible compared to the heat transfer through the thin film on the upper portion of the tube wall.
Rosson and Meyers [4] studied the stratified flow with higher vapor velocity using acetone and methanol as working fluids. They obtained the heat transfer coefficient at different angles around the periphery of the condenser tube and proposed different heat transfer correlations for top and bottom of the tube. For top of the tube, they modified the classic Nusselt model by introducing the effect of vapor shear. For the bottom of the tube, the von Karman analogy was used to predict the heat transfer coefficient.
After that, Shah [5] proposed a simple dimensionless correlation for predicting heat transfer coefficients during film condensation inside plane tubes. It was verified by comparison with a wide variety of experimental data. It was shown to agree with data for water, refrigerants, and organics covering a wide range of conditions in horizontal, vertical, and inclined tubes. However, in a later paper, the author stated that his correlation will fail at very low flow rates and high reduced pressure condition and the deviations were found to be related to the viscosity ratio of phases and reduced pressure. He improved and extended his correlation to a wider range of parameters [6] suggesting a correction factor that was developed through data analysis. This result was found to give good agreement with data at higher flow rates for both horizontal and vertical tubes.
Thome et al. [7] developed a flow pattern based on the condensation heat transfer model for the mist, annular, stratified wavy and fully stratified flow regimes. He proposed a new heat transfer model that

considers both film and convective condensation effects in the stratified wavy and fully stratified flow regimes. The correlation includes the effect of liquid-vapor interfacial roughness on heat transfer. They tried to obtain a method with a minimum of empirical constants and exponents that not only gives a good statistical representation of the data but also correctly captures the trends in data. Their model predicts the heat transfer coefficient within 20% error bound.
Cavallini et al. [8] also developed a condensation model for the annular flow and stratified flow regimes and validated it with over 2000 data points. Their stratified flow condensation model also includes both film and convective condensation heat transfer. Later, he proposed a new method [9] to determine the condensation heat transfer coefficient of fluids flowing into a horizontal smooth tubes with internal diameters D>3 mm. Although only two basic equations are proposed, one for higher flow rates (annular flow regime) and one for lower flow rates (wavy stratified flow regime), all possible flow regimes observed during the condensation process are included in their model. Their model was verified with a total number of 5478 data points relative to various refrigerants, carbon dioxide, ammonia, and water from several independent laboratories and showed a good agreement with experimental data.

Table I. The representative correlations for horizontal condensation heat transfer coefficient

Author	Correlation
Shah (1979)	$h_{TP} = h_L\left[(1-x^{0.8}) + \frac{3.8x^{0.76}(1-x)^{0.4}}{p_r^{0.38}}\right]$
Shah (2009)	$h_I = h_{LT}\left(\frac{\mu_f}{14\mu_g}\right)^n\left[(1-x^{0.8}) + \frac{3.8x^{0.76}(1-x)^{0.4}}{p_r^{0.38}}\right]$ $h_{NU} = 1.32\mathrm{Re}_{LS}^{-1/3}\left[\frac{\rho_f(\rho_f-\rho_g)gk_f^3}{\mu_f^2}\right]^{1/3}$
Cavallini (2002)	$h_D = \left[h_A\left(J_G^T/J_G\right)^{0.8} - h_{strat}\right]\left(J_G/J_G^T\right) + h_{strat}$
Rosson and Myer (1965)	$h_{top} = a\left(1+b\mathrm{Re}_g^c\right)\left(\frac{\mathrm{Gr}\mathrm{Pr}_f}{\mathrm{Ja}}\right)^{0.25}\left(\frac{k_g}{D_{hg}}\right)$ $h_{bottom} = 0.023\mathrm{Re}_f^{0.8}\mathrm{Pr}_f^{0.4}\frac{k_f}{D_{hf}}$
Cavallini (2006)	$h_D = \left[1-\left(J_G/J_G^T\right)\right]h_{strat}$ $h_L = 0.023\mathrm{Re}_f^{0.8}\mathrm{Pr}_f^{0.4}\frac{k_L}{D}\left(\frac{T_w-T_i}{T_w-T_{sat}}\right)$
Thome (2003)	$h_f = 0.728\left[\frac{\rho_L(\rho_L-\rho_V)gh_{LV}k_L^3}{\mu_L d(T_{sat}-T_w)}\right]^{1/4}$ $h_c = 0.003\mathrm{Re}_L^{0.74}\mathrm{Pr}_L^{0.5}\frac{k_L}{\delta}f_i$

From the above correlations, there are two kinds of method in modeling the condensation heat transfer phenomenon inside a nearly horizontal heat exchanger tube. One method is that the condensation phenomenon from the inlet to the outlet of the tube is modeled with one correlation. In the other method, the correlation is composed of two parts, convection and condensation heat transfer, separately. Shah's correlation [9] is also divided into two parts, convection and condensation heat transfer. The Shah's correlation uses the following two heat transfer equations;

$$h_I = h_{LT}\left(\frac{\mu_f}{14\mu_g}\right)^n \left[(1-x^{0.8}) + \frac{3.8x^{0.76}(1-x)^{0.4}}{p_r^{0.38}}\right] \tag{1}$$

$$n = 0.0058 + 0.557p_r \tag{2}$$

The second equation is

$$h_{NU} = 1.32\mathrm{Re}_{LS}^{-1/3}\left[\frac{\rho_f(\rho_f-\rho_g)gk_f^3}{\mu_f^2}\right]^{1/3} \tag{3}$$

Equation (3) is the Nusselt equation for laminar film condensation in vertical tubes. For horizontal tubes, the total condensation heat transfer coefficient is given as follows:

$$h_{TP} = h_I + h_{NU} \tag{4}$$

Precise modeling of heat transfer at the interface between each phase will contribute to reduce the difference between the correlation and experimental data. The modified correlation is used the slip ratio between each phase estimated from the thickness of the condensate water in place of the ratio representing the phase properties. In the present study, the ratio of dynamic viscosity between each phase is replaced with a slip ratio in Equation (1). To develop the slip ratio correlation using the condensate water level or the void fraction, it is assumed that a stratified steam-water mixture is flowing between two horizontal flat plates. Both phases are also assumed in fully steady state condition. The two-phase flow is shown in Fig. 1, where δ represents the thickness of the liquid layer.

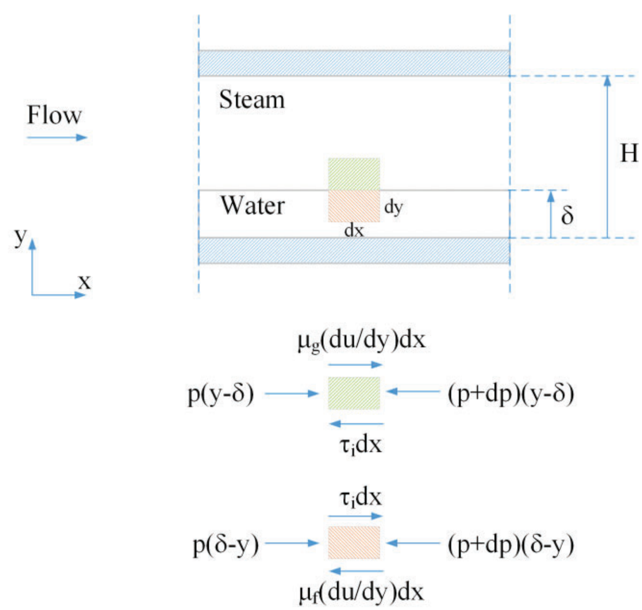

Figure 1. Physical model for condensation between horizontal plates.

A mass balance on the control volume shown in Fig. 1 yields the following expressions for each phase:

$$v(x,y) - v(x,\delta^-) = \int_y^\delta \frac{\partial u}{\partial x} dy \qquad \text{For } 0 < y < \delta \tag{5}$$

$$v(x,y) - v(x,\delta^+) = -\int_\delta^y \frac{\partial u}{\partial x} dy \qquad \text{For } \delta < y < H \tag{6}$$

For the fully develop flow, the terms on the right hand sides of the Equation (5) and (6) are set equal to zero. Neglecting any mass transfer provides the boundary condition. Consequently, the vertical component of the velocity field is equal to zero in the domain. Momentum transport in the flow direction yields the following equations for each phase, respectively:

$$0 = -\frac{dp}{dx}(\delta - y)dx + \tau_i dx - \mu_f \frac{du}{dy} dx \qquad \text{For } 0 < y < \delta \tag{7}$$

$$0 = -\frac{dp}{dx}(y - \delta)dx - \tau_i dx + \mu_g \frac{du}{dy} dx \qquad \text{For } \delta < y < H \tag{8}$$

The interfacial shear τ_i is related to the velocity gradients in both phases by

$$\tau_i = \mu_f \left.\frac{du}{dy}\right|_{\delta^-} = \mu_g \left.\frac{du}{dy}\right|_{\delta^+} \tag{9}$$

From Equation (7) and (8), the velocity profiles for the two phases are given by integrating the two ordinary differential equations by separating of variables and applying the no-slip condition at the wall:

$$u(y) = \frac{1}{2\mu_f}\frac{dp}{dx}[(\delta - y)^2 - \delta^2] + \frac{\tau_i}{\mu_f} y \qquad \text{For } 0 < y < \delta \tag{10}$$

$$u(y) = \frac{1}{2\mu_g}\frac{dp}{dx}[(y - \delta)^2 - (H - \delta)^2] + \frac{\tau_i}{\mu_g}(y - H) \qquad \text{For } \delta < y < H \tag{11}$$

The interface the two values of u must be equal and the equation is rearranged with $\delta = H(1-\alpha)$:

$$\frac{dp}{dx} = \left[\frac{2}{H}\frac{\mu_g(1-\alpha)+\mu_f\alpha}{\mu_g(1-\alpha)^2-\mu_f\alpha^2}\right]\tau_i \tag{12}$$

By integrating Equation (10) and (11), the average velocity of each phase is given as follows:

$$U_f = \frac{1}{\delta}\int_0^\delta \left\{\frac{1}{2\mu_f}\frac{dp}{dx}[(\delta - y)^2 - \delta^2] + \frac{\tau_i}{\mu_f} y\right\} dy = \frac{\delta}{2\mu_f}\left(\tau_i - \frac{2}{3}\frac{dp}{dx}\delta\right) \tag{13}$$

$$U_g = \frac{1}{H-\delta}\int_\delta^H \left\{\frac{1}{2\mu_g}\frac{dp}{dx}[(y - \delta)^2 - (H-\delta)^2] + \frac{\tau_i}{\mu_g}(y - H)\right\} dy = \frac{H-\delta}{2\mu_g}\left(-\tau_i - \frac{2}{3}\frac{dp}{dx}(H - \delta)\right) \tag{14}$$

From the definition of the slip ratio, S and the derived expression for the pressure gradient:

$$S = \frac{U_g}{U_f} = -\frac{\mu_f}{\mu_g}\frac{\alpha}{1-\alpha}\left[\frac{\tau_i + \frac{2dp}{3dx}H\alpha}{\tau_i - \frac{2dp}{3dx}H(1-\alpha)}\right] \tag{15}$$

$$S = \left(\frac{\mu_f}{\mu_g}\right)^2 \frac{\alpha}{1-\alpha}\left[\frac{\frac{1}{3}\alpha^2+\frac{4\mu_g}{3\mu_f}\alpha(1-\alpha)+\frac{\mu_g}{\mu_f}(1-\alpha)^2}{\frac{\mu_f}{\mu_g}\alpha^2+\frac{4\mu_f}{3\mu_g}\alpha(1-\alpha)+\frac{1}{3}(1-\alpha)^2}\right] \tag{16}$$

Using the derived slip ratio, Equation (1) is modified as follows:

$$h_I = h_{LT}(mS)^n\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{P_r^{0.38}}\right] \tag{17}$$

For compensating interfacial heat transfer between the steam and condensed water, we adopted the interfacial roughness factor.

$$h_{LT} = 0.023\mathrm{Re}_{LT}^{0.8}\mathrm{Pr}_{LT}^{0.4}\frac{k_f}{\delta}f_i \tag{18}$$

The condensation heat transfer correlation derived in the chapter was compared with the experimental data acquired from the experiments introduced in following Chapter 3.

3. COMPARISON WITH EXPERIMENTAL DATA

In order to evaluate the developed model for the condensation heat transfer coefficient, the experimental data from the HOCO and PASCAL facility was utilized. The PASCAL facility includes two kinds of the condensation heat exchanger, a U-tube and a nearly horizontal tube.
To acquire more accurate measurement of the condensation heat transfer coefficient, a horizontal straight tube experiment was done in HOCO facility. The test with the horizontal straight tube focused only on the condensation phenomena inside the condensation tube. The inner diameter and the thickness of the straight tube was the same with those of the prototype PCHX. The horizontal straight tube was installed inside a water jacket, which removes the heat from the horizontal straight tube using cooling water. The tube length was 2 m, which was similar to the straight region of the PCHX in the prototype. At only one location along the tube, the wall and fluid temperatures were measured by installing thermocouples inside the tube. Also, coolant temperature of the water jacket system was measured at the same locations, which enabled to calculate the heat transfer rate of the tube and the thermal equilibrium quality inside the tube. More details about the design for the straight tube test can be referred in the database report [10]. Since the major objective of the PASCAL test is to validate heat removal capability of the PCHX (Passive Condensation Heat Exchanger) in the APR+ PAFS, the U-tube condensation heat exchanger in the PASCAL facility has been designed to simulate equivalent phenomena inside the PCHX of the PAFS. It has the same length, inclination angle (3°), thickness (3 mm), diameter and material with the prototype PCHX. The PASCAL facility includes two kinds of the condensation heat exchanger, a U-tube and a nearly horizontal tube.
Table II and Table III summarized the test matrix for the condensation heat transfer coefficient in the straight tube and the PASCAL facility [11,12], respectively. The straight tube experiment (HOCO) was done using ultrasonic technique to measure the thickness of condensate water at the condition of low pressure and low temperature in which low flow rate of the steam was condensed. At each test, the data for the heat transfer coefficient was acquired during a steady state condition, when the steam was supplied from the steam boiler or generator with a constant pressure, temperature and flow rate.
The condensation heat transfer coefficient correlation modified with Equation (16) was compared with the experimental data acquired from the horizontal straight tube and the PASCAL experiment. The total number of data used for the evaluation was 82 except inlet, outlet and bending points of the PASCAL experimental data. The range of flow conditions for the horizontal straight tube experiment and the PASCAL experiment are summarized in Table IV. The vapor quality was assumed to vary linearly

through the whole test section for PASCAL experimental data. In Fig. 2, the experimental data are shown in a heat transfer regime map based on the criterion proposed by Shah [9].

Table II. Test matrix for the horizontal straight tube experiment

Test ID	System Pressure (kPa)	System Temperature (°C)	Mass flow rate (kg/s)
PCHX-QM-PT-01	101.858	101.1	0.002467
PCHX-QM-PT-02	101.950	100.3	0.003338
PCHX-QM-PT-03	101.154	100.1	0.003507
PCHX-QM-PT-04	100.273	100.0	0.003876
PCHX-QM-PT-05	102.734	100.6	0.005355
PCHX-QM-PT-06	103.407	100.7	0.005492
PCHX-QM-PT-07	104.170	100.9	0.006155
PCHX-QM-PT-08	102.836	101.2	0.006302
PCHX-QM-PT-09	105.437	103.7	0.007890
PCHX-QM-PT-10	106.758	102.4	0.008611
PCHX-QM-PT-11	112.402	103.3	0.009477
PCHX-QM-PT-12	115.345	104.1	0.010033
PCHX-QM-PT-13	119.521	105.1	0.010739

Table III. Test matrix for the PASCAL experiment

Test ID	System Pressure (MPa)	System Temperature (°C)	Mass flow rate (kg/s)
SS-200-P1	0.85	175.2	0.0937
SS-300-P1	1.34	194.6	0.1469
SS-400-P1	1.97	213.0	0.2037
SS-540-P1	3.22	239.1	0.2953
SS-650-P1	4.70	261.3	0.3651
SS-750-P1	6.74	284.4	0.4302
PASCAL-350-01	1.51	203.28	0.1676
PASCAL-450-01	2.22	222.47	0.2264
PASCAL-500-01	2.61	230.82	0.2565
PASCAL-600-01	3.74	250.49	0.3107
PASCAL-700-01	5.66	275.26	0.3661
PHCX-400-00	5.74	275.69	0.1054
PHCX-315-01	0.58	159.18	0.0393
PHCX-224-01	4.36	258.43	0.1132

PHCX-250-01	7.29	291.59	0.1420
PHCX-100-01	0.28	130.83	0.0366
PHCX-120-01	0.39	143.22	0.0431
PHCX-140-01	0.66	164.07	0.0537
PHCX-160-01	1.23	191.25	0.0692
PHCX-180-01	2.05	216.31	0.0848
PHCX-200-01	3.28	241.66	0.1012
PHCX-270-01	1.09	185.72	0.0732
PHCX-300-01	1.36	196.10	0.0807
PHCX-350-01	2.00	215.01	0.0948
PHCX-400-01	2.98	236.12	0.1116
PHCX-450-01	4.26	256.92	0.1307
PHCX-270-02	1.97	214.16	0.1009
PHCX-300-02	2.57	228.01	0.1133
PHCX-350-02	3.83	250.72	0.1350
PHCX-400-02	5.71	275.33	0.1618
PHCX-450-02	7.24	291.04	0.1831

Table IV. Flow Conditions for the horizontal straight tube experiment

Source	Reduced pressure, p_r	Mass velocity (kg/m^2s)	Vapor quality	Tube inner diameter (m)	Film thickness (mm)	Case #
Horizontal straight tube experiment	0.0046~0.0054	1.56~6.81	0~1	0.0448	5.07~7.76	13
PASCAL experiment	0.038 ~ 0.328	46 - 273	0~1	0.0448	10.5~18.6	69

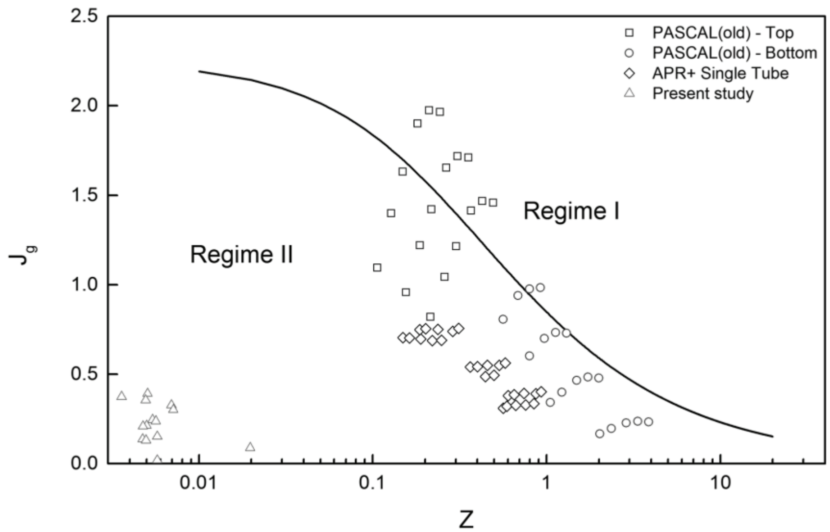

Figure 2. Heat transfer regime for the horizontal straight tube experiment and the PASCAL experiment.

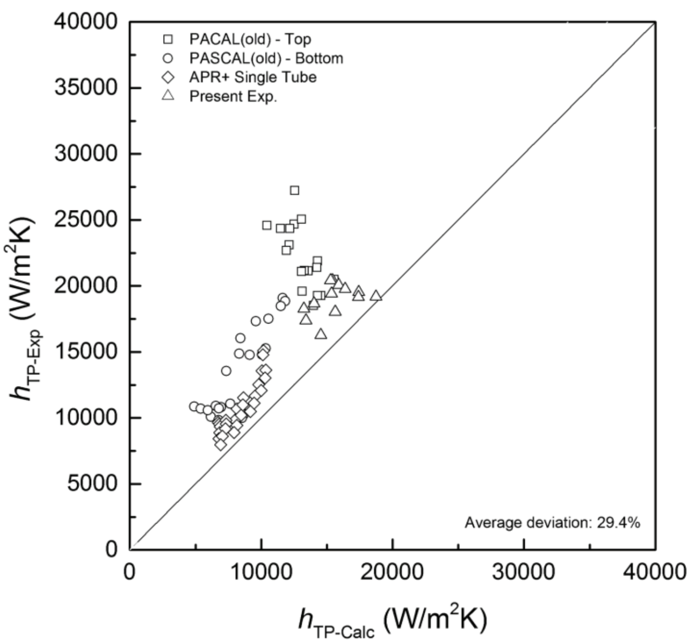

Figure 3. Total condensation heat transfer coefficient using Shah (2009) correlation.

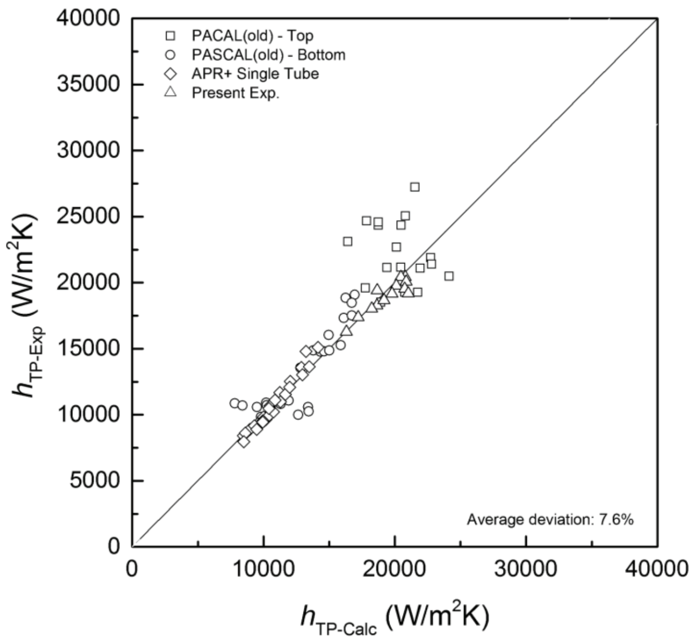

Figure 4. Total condensation heat transfer coefficient using the proposed correlation.

In the horizontal straight-tube experimental data, the quality was derived from the heat balance calculation between the condensation heat exchanger tube and the water jacket. The total condensation heat transfer coefficient based on Shah [9] correlation was compared with the experimental data shown in Fig. 3. The heat transfer coefficients showed dispersed trend and underestimated compared to the experimental data. The percentage error from the comparison with the experimental data was about 30%. Figure 4 shows a comparison of the modified total condensation heat transfer coefficient and the experimental data. The constants m and n of Equation (17) were selected to minimize the error from the comparison with the horizontal straight-tube experiment except the PASCAL data. Then, the modified condensation heat transfer coefficient correlations using the developed slip ratio and interfacial roughness factor are as follows:

$$h_I = h_{LT}(0.52S)^{-0.173}\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{P_r^{0.38}}\right] \tag{19}$$

$$h_{LT} = 0.023\mathrm{Re}_{LT}^{0.8}\mathrm{Pr}_{LT}^{0.4}\frac{k_f}{\delta}\left(\frac{\rho_f}{\rho_g}\right)^{1/3} \tag{20}$$

The percentage error from the comparison with the experimental data is about 8%. Throughout this result, the derived slip ratio model for developing the heat transfer coefficient correlation in the condensation heat exchanger tube would appear quite reasonable method.

4. CONCLUSIONS

The condensation heat transfer coefficient correlation for the horizontal heat exchanger was developed to model the heat transfer phenomenon inside the horizontal heat exchanger. The modified condensation heat transfer coefficient correlation showed improved performance for predicting heat transfer coefficient within 8% error bound. To increase the prediction capability of the condensation heat transfer coefficient correlation developed in this study, various pressure and temperature conditions inside the condensation tube can be conducted as the further study.

NOMENCLATURE

C_P	Specific heat at constant pressure [J/kgK]
G	Total mass flux (liquid + vapor)
H	Tube inner diameter [m]
g	Gravity acceleration [m/s^2]
h	Heat transfer coefficient [W/m^{2}K]
h_I	Heat transfer coefficient given by Equation (1)
h_{LS}	Heat transfer coefficient assuming liquid phase flowing alone in the tube
h_{LT}	Heat transfer coefficient assuming all mass flowing as liquid
h_{Nu}	Heat transfer coefficient given by Equation (3)
h_{TP}	Two-phase heat transfer coefficient
k	Thermal conductivity [W/mK]
Nu	Nusselt number [hD/k]
Re	Reynolds number [ρuD/μ]
Pr	Prandtl number [$\mu C_p/k$]
p_r	Rreduced pressure
Re_{LS}	Reynolds number assuming liquid phase flowing alone [$G(1-x)H/\mu_f$]
T_{sat}	Saturation temperature [K]
T_w	Wall temperature [K]
U	Velocity [m/s]
x	Vapor quality

Greek Symbols

α	Void fraction
δ	Liquid film thickness
μ	Dynamic viscosity [N-s/m^2]
ρ	Density [kg/m^3]

Subscripts

g	Gas phase
f	Liquid phase

ACKNOWLEDGMENTS

This work is supported by the National Research Foundation of Korea (NRF- 2012M2A8A4004176) grant funded by Ministry of Science, ICT and Future Planning of the Korea government.

REFERENCES

1. Park Y.S. et al., “Report on the Model Package Development of the Condensation Heat Transfer for PAFS Best-estimation Analysis,” APR+-PAFS-TR-14-01 (2014).
2. Chato, J.C., “Laminar condensation inside horizontal and inclined tubes.” *ASHRAE J.*, **4**(2), pp. 52–60 (1962).
3. Nusselt, W., “Die Oberflachenkondensation des Wasserdampfes (The surface condensation of water vapor),” *Zeit. D. Ver. Deut. Ing. Frankfurt*, **60**, pp. 541–575 (1916).
4. Rosson, H.F., Meyers, J.A., “Point of Values of Condensing Film Coefficients inside a Horizontal Tube,” *Chem. Eng. Prog. Symp. Ser.*, **61**, pp. 190–199 (1965).

5. Shah, M.M., “A General Correlation for Heat Transfer during Film Condensation inside Pipes,” *Int. J. Heat Mass Transfer*, **22**, pp. 547–556 (1979).
6. Shah, M.M., “An Improved and Extended General Correlation for Heat Transfer during Condensation in Plain Tubes,” *HVAC&R RESEARCH*, **15**(5), pp. 889-913 (2009).
7. Thome, J.R., El Hajal, J., Cavallini, A., “Condensation in Horizontal Tubes. Part 2: New Heat Transfer Model based on Flow Regimes,” *Int. J. Heat Mass Transfer*, **46**, pp. 3365–3387 (2003).
8. Cavallini, A., Censi, G., Del Col, D., Doretti, L., Longo, G.A., Rossetto, L., “Condensation of Halogenated Refrigerants inside Smooth Tubes,” *Int. J. HVAC&R Res.*, **8**(4), pp. 429–451 (2002).
9. Cavallini, A., Del Col, D., Doretti, L., Matkovic, M., Rossetto, L., Zilio, C., Censi, G., “Condensation in Horizontal Smooth Tubes: a New Heat Transfer Model for Heat Exchanger Design,” *Heat Transfer Eng.*, **27**(8), pp. 31–38 (2006).
10. Kim S. et al., “Database Report of Condensation Heat Transfer Experiment for Horizontal Condensation Heat Exchanger,” KAERI/TR-5845/2015 (2015).
11. B.J. Yun et al., “Construction Report of Separate Effect Test Facility for Passive Auxiliary Feedwater System (PASCAL),” KAERI/TR-4085/2010 (2010).
12. B.U. Bae et al., “Construction Report of the Separate-effect Test Facility to Develop Model Package for Condensation Heat Transfer,” APR+-PAFS-DD-13-01 (2013).